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INFLUENCE OF HEAT EXCHANGER EFFECTIVENESS AND SYSTEM FLOW RATES ON EXPERIMENTAL RATINGS OF A GENERIC ANTIFREEZE SDHW SYSTEM

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ABSTRACT

Effects of heat exchanger design, collector-loop flow rate, and recirculation flow rate on the thermal rating of a generic antifreeze solar domestic hot water heating system are experimentally investigated in 11 Solar Rating and Certification Corporation OG-200 rating trials. Collector-loop flow rates are varied from 0.057 l/s to 0.228 l/s, and recirculation flow rates from 0.043 l/s to 0.176 l/s. Two double-walled heat exchanger designs are tested: a counter-flow U-shaped tube-in-tube with micro-fins and a counter-flow 8-pass tube-in-tube without fins. Heat exchanger effectiveness varies from 0.164 to 0.343. Collector area and tank volume are held constant at 5.56 m^2 and 246 l, respectively. Over the ranges examined, system performance is insensitive to changes in heat exchanger effectivenesses above 0.17. In the absence of stratification, lower recirculation flow rates are not advantageous. Reductions in collector flow rate cause higher collector operating temperatures and reduced thermal output without any benefit to heat exchanger performance.

1. INTRODUCTION

Performance of a solar domestic hot water (SDHW) system is strongly dependent on operating parameters and component specifications. Collector performance and size have the largest impact on system energy output. However, tank stratification, parasitic power, and heat exchanger effectiveness, ϵ, can also influence system temperatures and overall efficiency. Antifreeze provides excellent protection against freezing and scale build-up. However, in many locations, its use requires the installation of a double-walled heat exchanger. In this case, lowered heat exchanger effectiveness may degrade system performance.

The Solar Rating and Certification Corporation (SRCC) OG-200 (1) rating test is used to study performance of a generic antifreeze (50%) SDHW system in a parametric study of system flow rates and heat exchanger design. Eleven rating trials are conducted for a fixed collector area of 5.56 m^2 and a constant tank volume of 246 l. Collector-loop flow rates are varied from 0.057 l/s to 0.228 l/s. The collector manufacturer recommends a flow rate of 0.114 l/s for this collector area. Recirculation flow rates are varied from 0.043 l/s to 0.170 l/s. Conventional water flow rates are normally 0.19 to 0.25 l/s. Two double-walled heat exchanger designs are tested: 1) a counter-flow U-shaped tube-in-tube with micro-fins and a heat exchange area of 0.214 m^2, and 2) an 8-pass counter-flow tube-in-tube with heat exchange area of 0.279 m^2. The U-tube heat exchanger is also tested in an inverted orientation with the inlets at the bottom.

2. EXPERIMENTAL FACILITY

The SDHW rating facility is located in one of five 2-story residential style solar heated and cooled buildings at the Solar Energy Applications Laboratory. Water tanks, pumps, and heat exchanger are placed at ground level. Collectors are positioned at a 45 degree angle in a darkened, climate controlled space between an outer false roof and a weather-tight inner house roof. Solar radiation input is simulated by an in-line electric heater located downstream of the collectors. A detailed description of the facility is included in Carlson (2).

System control (including pumps, electric heater, and water draws) and data acquisition and analysis are integrated around a 80386-based personal computer system. Temperatures, temperature differences, flow rates and parasitic power are monitored throughout the facility as shown in Figure 1. Temperatures are measured with copper-constantan thermocouples (± 1 deg C) and temperature differences are determined with 5-junction thermopiles ($\pm$(1% of reading + 0.05 deg C)). Volumetric fluid flow rates are measured with turbine meters ($\pm$0.5% linearity, standard error $\leq$0.001 l/s). Mass flow rates for water draws are also measured with turbine flow meters (standard error $\leq$0.0002 kg/s). Electrical energy consumption is measured using Watt transducers.

A description of the SDHW system is given in Table 1. Pump sizes vary due to higher pressure drops across the 8-pass heat exchanger at higher flow rates.

TABLE 1 COMPONENT SPECIFICATIONS

Component	Specification	Value	Notes
Collector loop fluid	50/50 ethylene glycol	Property values from ASHRAE Fundamentals	
Collector	Total Area	5.56 m^2	
	$F_r\tau\alpha$	0.602	
	F_rU_l	5.55 $W/m^2/K$	
	b_o	0.42	
	$K_{\tau\alpha}$ @ θ = 0	1.00	
	15	0.97	
	30	0.90	
	45	0.75	
	60	0.58	
	mC_p/Area (test)	74.0 $W/m^2/K$	1
Solar Storage Tank			
	Volume	246 liter	2
	R-Value	2.10 C m^2 /W	
Auxiliary DHW	Volume	159 liter	2
	R-Value	2.10 C m^2/W	2
	Set Point	55 C	
Heat Exchanger			
1	Counter flow, copper, double wall, micro-finned		
	Heat exchange area	0.214 m^2	2
2	Counter flow, 8-pass, copper, double wall, no fins		
	Heat exchange area	0.279 m^2	2
Pumps: Collector	Type	Grunfus UP 15-42-F Nominal 85 W	2,4
		Grunfus UP 26-96F Nominal 205 W	5,6
Recircu-lation	Type	Grunfus UP 15-18SU Nominal 85 W	2,4
		Grunfus UP 15-42-F Nominal 85 W	5
		Grunfus UP 26-96F Nominal 205 W	6
Piping	Size and Type		
	Insulation Type	Closed Cell Foam 1.9 cm	
	Insulation R-Value	0.82 C m^2/W	
	Lengths:		
	Heat Exchanger to Collector	10.82 m	3
	Collector to Loop Heater	1.39 m	3
	Loop Heater to Heat Exchanger	11.71 m	3
	Heat Exchanger to Solar Storage Tank	2.90 m	3
	Solar Storage Tank to Heat Exchanger	3.15 m	3
Valves	Type	Gate	
	Size	19 mm	

Notes
1. Value pertains to time of collector rating test.
2. Manufacturer's rating.
3. Value is based on measurements made in laboratory.
4. Used in all Trials with HX 1, ie. Trial 1-4, and 9-10; and Trials 6 and 11 with HX 2.
5. Used in Trial 8
6. Used in Trials 5 and 7

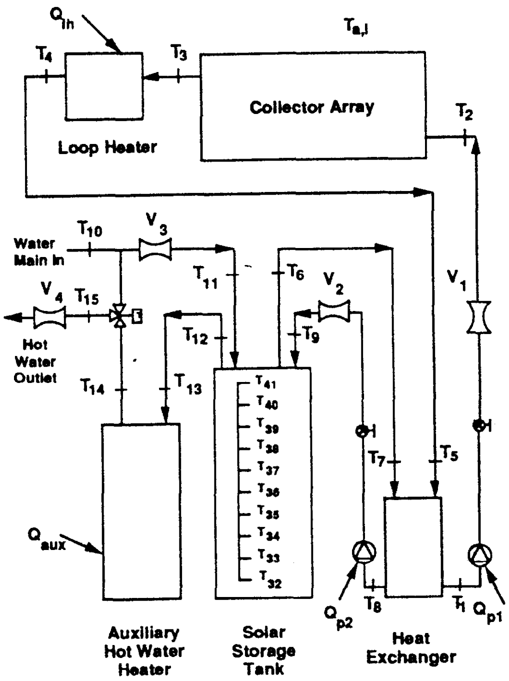

Fig. 1. Schematic of Antifreeze System

3. METHODOLOGY

The SRCC short-term rating procedure evaluates the steady-state performance of a SDHW system during four days. Environmental conditions and water load demands are specified as shown in Table 2. Daily hot water load is based on three equal draws at 8:00, 12:00, and 17:00. In these rating trials, the specified daily water draw is increased from 42.3 MJ to 49.8 MJ to conform with load specifications for conventional DHW systems (3). The in-line heater input is controlled according to the specified hourly solar profile and calculations outlined in the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 95-1987 (4). Collector characteristics are determined in a separate ASHRAE Standard 93-1986 test (5).

TABLE 2 SRCC RATING SPECIFICATIONS

Quantity	Specification
Ambient Temperature	22 ±2 deg C
Mains Water Temperature	22 ±1 deg C
Total Daily Insolation	17.03 MJ / m^2
Total Daily Hot Water Load	42.3 MJ
Minimum Hot Water Temperature	35 deg C
Hot Water Draw Flow rate	0.2 kg/s
Auxiliary Heater Set Point	48.9 deg C

A schematic of most of the SRCC energy quantities is shown in Figure 2. Energy delivered by the solar collectors is Q_u. Energy drawn from the solar storage tank, Q_s, minus the parasitic energy used by controllers and pumps, Q_{par}, is Q_{net}. Daily energy input into the auxiliary hot water heater is Q_{aux}. Daily energy delivered to the load is Q_{del}. Reserve energy left in the solar storage tank at the end of the test is Q_{res}. Capacity of the system, Q_{cap}, is the amount of energy the system can deliver without solar input.

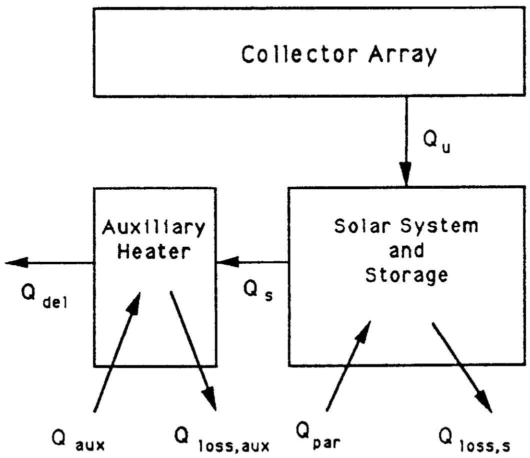

Fig. 2. Schematic of SRCC Energy Quantities

A system rating test is completed when the daily value of auxiliary energy, Q_{aux}, is within 3% of the previous day's value. The test is terminated after four days if convergence is not obtained. System thermal rating is based on daily energy quantities measured on the last day of testing if convergence is achieved. If a test does not converge, the average of daily quantities determined during the last two days is used to establish system rating. Daily energy quantities are calculated from measured temperature differences and flow rates as shown in the Appendix.

Heat exchanger effectiveness, the ratio of actual heat transfer across the heat exchanger to the maximum heat transfer possible between the hot and cold fluids, is given by,

$$\epsilon = \frac{\{\rho(T_8)C_p(T_8)V_2(T_8-T_7)\}}{\{[\rho(T)C_p(T)V]_{min}(T_5-T_7)\}} \qquad (1)$$

Number subscripts refer to the transducer locations shown in Figure 1. Effectiveness is calculated every fifteen minutes and averaged to provide a daily ϵ. Instabilities in ϵ that occur at the beginning and the end of the day are not included in the daily average.

4. RESULTS

Daily energy values and average daily heat exchanger effectivenesses are listed in Table 3 along with the flow rates and heat exchanger used in each of the 11 rating trials. The collector-loop antifreeze mixture is in the inner tube in the U-configuration and in the outer tube in the inverted position. The antifreeze flow is always in the outer tube of the 8-pass heat exchanger.

It is apparent in all 11 trials that use of a double-walled heat exchanger reduces heat exchanger effectiveness. In comparison to daily values of $\epsilon < 0.35$, effectiveness of a simple in-tank coiled copper tube heat exchanger, used in SRCC tests of a generic drain back SDHW system, never dropped below 0.40 and was as high as 0.60 (2).

Daily values of useful collected energy, Q_u, energy delivered from the solar storage tank, Q_s, and net energy delivered from the solar system, Q_{net}, are plotted as functions of heat exchanger effectiveness in Figures 3, 4 and 5, respectively. Data points are labeled by trial number. Measurement errors are calculated for each trial by combining transducer errors using a root sum

TABLE 3 SUMMARY OF EXPERIMENTAL RESULTS

Trial	Heat Exchanger Type	Collector Flow Rate [l/s]	Recirculation Flow Rate [l/s]	Effectiveness	Solar Fraction [%]	Qs [MJ]	Qnet [MJ]	Qres [MJ]	Qaux [MJ]	Qpar [MJ]	Qu [MJ]
1	1	0.114	0.170	0.308	41.2	24.8	20.5	18.0	29.9	4.3	31.1
2	1	0.114	0.085	0.266	39.6	24.0	19.7	17.3	30.6	4.2	30.0
3	1	0.228	0.170	0.242	44.3	26.4	22.0	19.6	28.7	4.4	31.2
4	1	0.228	0.085	0.324	41.7	25.2	20.8	18.2	29.7	4.4	29.7
5	2	0.114	0.170	0.226	36.2	24.3	18.0	18.5	31.8	6.3	29.3
6	2	0.114	0.085	0.343	42.0	25.2	20.9	18.0	29.8	4.3	30.6
7	2	0.228	0.170	0.180	37.8	25.5	18.8	19.1	30.1	6.6	30.0
8	2	0.228	0.085	0.290	34.6	23.6	17.3	18.1	31.8	6.3	29.9
9	1*	0.114	0.176	0.164	33.2	20.9	16.6	16.1	34.3	4.3	28.1
10	1*	0.114	0.095	0.167	33.8	21.1	16.8	15.8	33.8	4.3	27.8
11	2	0.057	0.043	0.316	36.1	22.4	18.0	15.1	32.6	4.4	28.5

* Inverted Heat Exchanger: Inlets are at the bottom

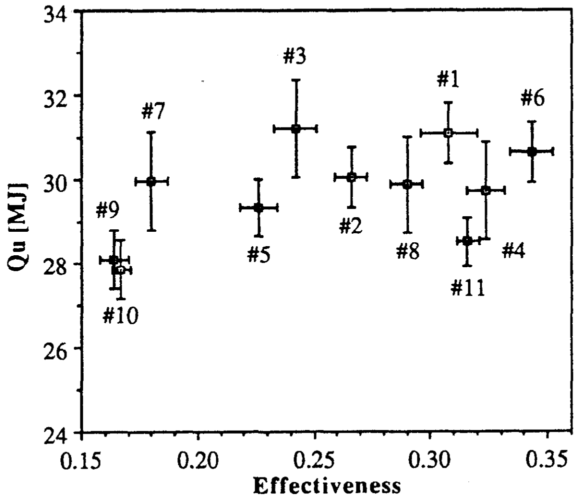

Fig. 3. Useful Collected Energy as a Function of Effectiveness.

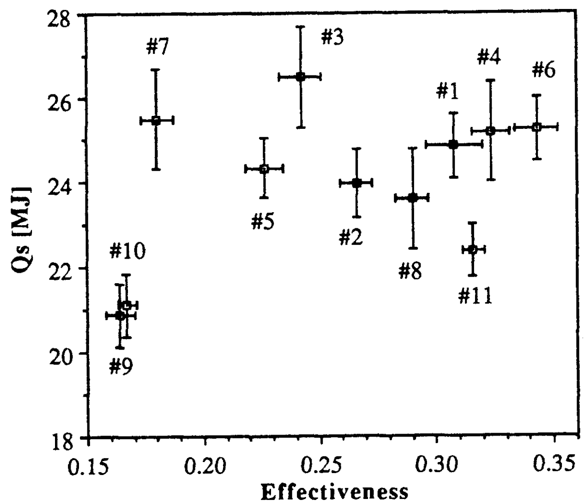

Fig. 4. Solar Energy Supplied to Load as a Function of Effectiveness.

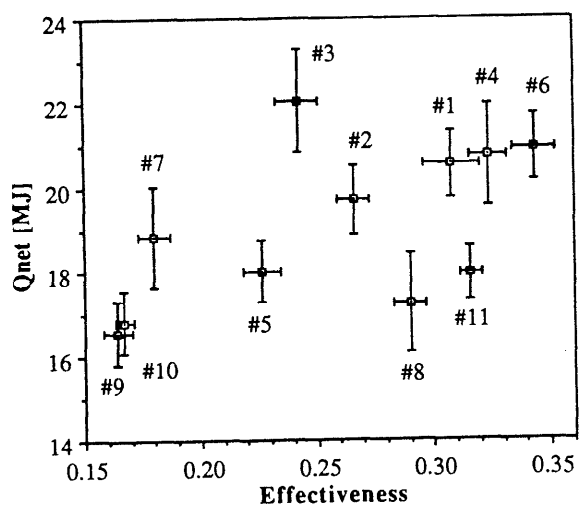

Fig. 5. Net Solar Energy Supplied to Load as a Function of Effectiveness.

square method. Experimental error is assumed to be ± two standard deviations. Error bars for ϵ are daily averages of the 15 minute instantaneous errors. Caution must be taken in interpreting net energy delivered. Although this quantity is calculated by subtracting the thermal equivalent of electrical input to the two pumps from Q_s, only a portion of the pumping power is returned to the fluid as heat. Thus, in trials using larger pumps, i.e. Trials 5, 7 and 8, Q_{net} is reduced disproportionally compared to the other trials.

Only in Trials 9 and 10, in which effectivenesses are on the order of 0.17, is there a drop in energy output outside the error bands of the other trials. Low values of ϵ in these inverted heat exchanger trials are probably caused by air trapped at the top of the tube. Otherwise, the data do not show a strong correlation between ϵ and Q_u, Q_s or Q_{net} over the range of variables tested. However, a comparison of Trials 5 and 9, which have equivalent flow rates, shows that a 27% decrease in ϵ from 0.226 to 0.164 causes a 4% decrease in Q_u and a 14% decrease in Q_s. This trend is supported by comparison of Trials 2 and 10, which also have similar flow rates. With a 37% decrease in ϵ from 0.266 to 0.167, Q_u decreases by 7% and Q_s decreases 12%.

Reducing the recirculation flow rate from 0.170 l/s to 0.043 l/s does not affect tank stratification. With the lowest recirculation flow rate, Trial 11 has the greatest potential for stratification, yet the plot of tank temperatures, shown in Figure 6, indicates that the tank is fully mixed except during night hours. The lag in water temperature at the bottom of the tank (T32) results from incomplete circulation of the tank volume due to location of the drop tube 0.06 m above the bottom of the tank. Without tank stratification, there is no advantage to reducing storage-side flow rate unless pumping power can be reduced significantly.

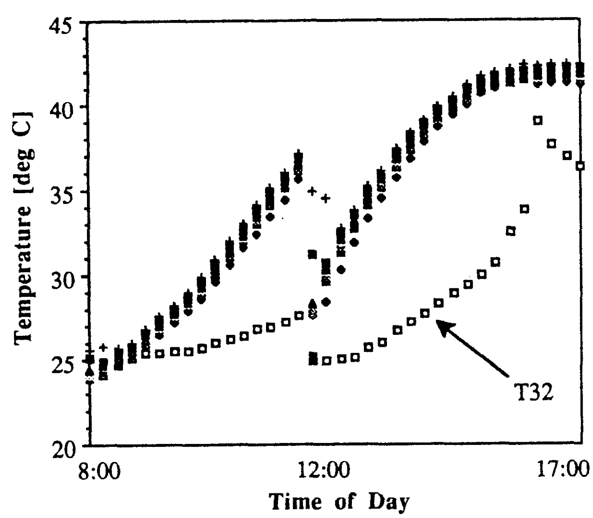

Fig. 6. Solar Storage Tank Temperatures for Trial 11.

The disadvantage of low flow rates when stratification is not achieved is evident in the Trial 11 results. Despite a relatively high average ϵ, this system is inferior to all the other configurations except those of Trials 9 and 10 in which the inverted heat exchanger is used. In Figure 7, comparison of collector inlet temperatures in Trials 1 and 11 with nearly equal values of ϵ but different collector-loop flow rates shows that in Trial 1, the higher collector-loop flow rate reduces collector operating temperatures throughout the day.

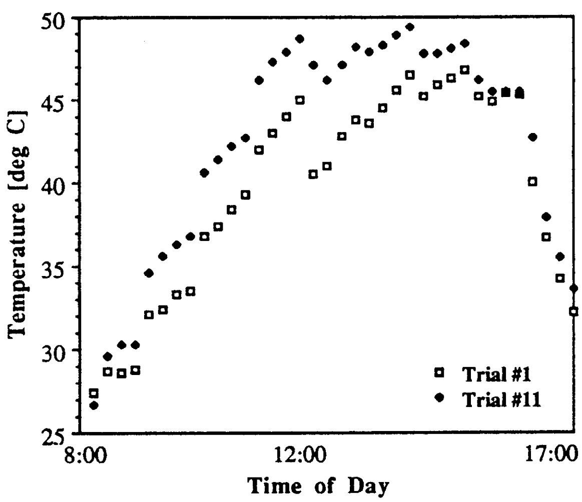

Fig. 7. Collector Inlet Temperatures of Trials 1 and 11.

5. CONCLUSION

In a generic antifreeze SDHW system, thermal output is insensitive to changes in heat exchanger effectiveness from 0.18 to 0.35. With equivalent flow rates, system performance is only degraded when effectiveness drops below 0.18. In the absence of stratification, higher collector and recirculation flow rates lead to better system performance, in spite of lower effectiveness.

6. ACKNOWLEDGEMENT

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7. NOMENCLATURE

b_o	=	dimensionless constant used in incident angle modifier calculation
C_p	=	specific heat, kJ/kg/K
F_r	=	collector heat removal factor
$K_{\tau\alpha}$	=	empirical incident angle modifier
m	=	mass flow rate, kg/s
Q	=	daily energy, kJ
T	=	temperature, deg C
t	=	time, s
UA	=	overall heat transfer coefficient, W/C
U_l	=	collector heat loss coefficient, $W/m^2/C$
V	=	volumetric flow rate, l/s

Greek Letters

θ	=	incident angle
ρ	=	water density, kg/m^3
$\tau\alpha$	=	transmittance absorptance product for collector
ϵ	=	heat exchanger

Subscripts

a	=	ambient
aux	=	auxiliary
cap	=	refers to energy capacity of system without solar input
del	=	refers to hot water energy delivered by system
dl	=	refers to desired hot water energy load on system
l	=	laboratory
loss	=	energy loss
lh	=	loop heater, refers to collector loop heater energy input
main	=	water main
net	=	net energy delivered from preheat tank
p1	=	collector loop pump, energy consumed by collector loop pump
p2	=	recirculation loop pump, energy consumed by recirculation loop pump
par	=	parasitic energy consumption
res	=	reserve energy in preheat tank at end of test sequence
s	=	solar, refers to hot water energy delivered form preheat tank
u	=	useful, refers to useful energy gain of fluid through collector array

8. REFERENCES

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(2) Carlson, W.T., Comparison of Experimental and TRNSYS SRCC Ratings of a Generic Drain Back Solar Water System, Master's Thesis, Colorado State University, Fort Collins Colorado (1990).

(3) Federal Trade Commission, Uniform Test Method for Measuring the Energy Consumption of Water Heaters, Code of Federal Regulations, 10 CFR Ch. II, Part 430, Subpart B, Appendix E (1989).

(4) American Society of Heating, Refrigerating and Air-Conditioning Engineers, Methods of Testing to Determine the Thermal Performance of Solar Domestic Water Heating Systems, Standard 95-1987, Atlanta, Georgia (1987).

(5) American Society of Heating, Refrigerating and Air-Conditioning Engineers, Methods of Testing to Determine the Thermal Performance of Solar Collectors, Standard 93-1986, Atlanta, Georgia (1986).

9. APPENDIX

The following equations are used to calculate the SRCC daily energy quantities.

$$Q_u = \int_{t=08:00}^{t=17:00} \rho(T_1)\, C_p(T_1)\, V_1\, (T_4 - T_2)\, dt \qquad \text{(A.1)}$$

$$Q_{del} = \int_{t=0}^{t=t(Q_{del}=Q_{d})} \rho(T_{15})\, C_p(T_{15})\, V_4\, (T_{15} - T_{10})\, dt \qquad \text{(A.2)}$$

$$Q_s = \int_{t=0}^{t=t(Q_{del}=Q_{d})} \rho(T_{11})\, C_P(T_{11}) V_3\, (T_{12} - T_{11})\, dt \qquad \text{(A.3)}$$

$$Q_{res} = \int_{t=0}^{t=t(T_{12} \leq T_{10}+3^{\circ}C)} \rho(T_{11})\, C_p(T_{11})\, V_3\, (T_{12} - T_{11})\, dt \qquad \text{(A.4)}$$

$$Q_{cap} = \int_{t=0}^{t=t(T_{15} \leq 35^{\circ}C)} \rho(T_{15})\, C_p(T_{15})\, V_4 (T_{15} - T_{11})\, dt \qquad \text{(A.5)}$$

$$Q_{net} = Q_{del} - Q_{aux} + Q_{loss} - Q_{par} \qquad \text{(A.6)}$$

$$= Q_s - Q_{par}$$

$$Q_{par} = Q_{p1} + Q_{p2} \qquad \text{(A.7)}$$